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Example to illustrate the use of Data Items on noise from ducted ventilation and air-conditioning systems

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THE PREPARATION OF THIS DATA ITEM

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The Internal Flow Panel, which took over the work on the Internal flow previously monitored by the Fluid Mechanics Steering Group, first met in September 1979 and has the co-operation of many engineers and scientists in industry, research establishments and universities from whom much assistance and information is being received.

EXAMPLE TO ILLUSTRATE THE USE OF DATA ITEMS ON NOISE FROM DUCTED VENTILATION AND AIR-CONDITIONING SYSTEMS

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EXAMPLE TO ILLUSTRATE THE USE OF DATA ITEMS ON NOISE FROM DUCTED VENTILATION AND AIR-CONDITIONING SYSTEMS

1. NOTATION*

		<i>SI unit</i>
A	free cross-sectional area of duct	m^2
A_G	free area of grille	m^2
B	damper blockage factor	–
b	duct height normal to damper axis	m
D_d	direct sound pressure level directivity correction	dB
D_r	direct sound pressure level distance correction	dB
d	duct diameter (for rectangular ducts, equivalent $d = \sqrt{4(A/\pi)}$)	m
d_c	characteristic dimension of duct cross section	m
F	flow noise spectrum function	dB
F_D	damper spectrum function	dB
ΔF_D	damper height correction	dB
F_G	grille spectrum function	dB
ΔF_G	grille vane thickness correction	dB
f	octave band centre frequency	Hz
f_B	fan blade passing frequency	Hz
G	flow velocity function	dB
H	octave bandwidth function	dB
K_w	fan specific sound power level	dB
k	grille constant	
L	sound level	dB

* The notation used in this Item is consistent with that used in other Data Items in the "Noise in air-conditioning systems" volume of the Fluid Mechanics, Internal Flow Sub-series.

ΔL	increment in sound level	dB
L_{pd}	direct sound pressure level	dB
L_{pr}	reverberant sound pressure level	dB
ΔL_{pr}	reverberant sound pressure level multiple grille correction	dB
L_w	octave band sound power level	dB
L_w^*	fan base sound power level	dB
ΔL_w	octave band attenuation	dB
P	perimeter inside duct	m
$\Delta p_{f,t}$	fan total pressure rise	Pa
Q	volume flow rate in duct	m ³ /s
q	volume flow rate through fan	m ³ /s
R_T	reverberant sound pressure level reverberation time correction	dB
R_v	reverberant sound pressure level room volume correction	dB
r	distance from centre of duct outlet to observation point	m
Sr	Strouhal number ($Sr = fd/V$)	–
Sr_D	damper Strouhal number ($Sr_D = fd/V_e$)	–
Sr_G	grille Strouhal number ($Sr_G = ft_G/V_e$)	–
Sr_T	transition Strouhal number ($Sr_T = fd/V$)	–
T	reverberation time	s
t_G	grille vane thickness	m
V	mean flow velocity	m/s
V_e	effective flow velocity	m/s
v	room volume	m ³
θ	damper vane angle	deg

2. INTRODUCTION

This Item illustrates, by means of a worked example, the application of data provided in the Data Items listed as References 1 to 4. These Items provide the essential data for the estimation of sound pressure levels resulting from the operation of a ducted ventilation system within a room served by that system. These sound pressure levels may then be checked against the required design noise criteria to assess the need for sound reduction within the air distribution system. It is unlikely that any ventilation system will meet the required limiting noise criteria without the addition of some sound reducing treatment. The example calculations in this Item should be used in conjunction with those in ESDU 82001 and 82002 illustrating the estimation of the reverberation time for an enclosure, and the selection and sizing of a silencer for a ventilation system. The Item is intended for use by engineers who are not acoustic specialists.

3. EXAMPLE

It is required to investigate the sound pressure levels, due to the air conditioning and ventilation system, in the offices shown in Figure 1 and compare the levels with those permitted by the required limiting noise criteria. The permitted noise ratings, for each room within the office complex, are given on the Figure. These noise ratings are consistent with the typical noise limiting indices in Table 9.2 of ESDU 82001. The ducts are made of sheet steel and are rectangular in section, duct sizes being given in mm units in parentheses on Figure 1.

For the purpose of this example, it may be assumed that the rooms served by the second branch have less critical limiting noise criteria than the branch shown in Figure 1. Therefore the second branch need not be considered in the noise calculations. The required air volume flow rates have been estimated for each room and the flow velocities, shown in Figure 1, calculated from the given duct sizes. To achieve the required flow rates in all the branches additional flow control dampers may be needed but are omitted here to avoid complicating the example calculations.

Only the air delivery system is shown on the Figure but an allowance for the extract system is to be made, where appropriate, in the calculations. The duct outlets serving the corridor area supply relatively small quantities of air and may be neglected in the noise calculations.

3.1 Presentation of Example Calculations

It is necessary to consider the noise levels in each of the eight octave frequency bands from the band centred at 63 Hz through to the band centred at 8 kHz. The calculations are presented in tabular form, and to show the details of these calculations and the source of data used at each step, the calculation for a single octave frequency band is also presented. The detailed calculations are referenced from the appropriate line of the tabulated calculation.

It is convenient to consider the propagation, through the duct system, of the fan noise and the flow generated noise separately since it is then easier to estimate the effects of noise reduction on the system. Silencers are usually located near the fan (see ESDU 82002, Section 5) and are therefore only effective in attenuating the noise generated by the fan and other items located before the silencer. Separating the calculation for the two noise sources allows easy comparison of the silencer attenuated fan sound levels with the flow generated noise within any serviced space. If, after silencing the fan, the flow generated noise significantly contributes to, or dominates, the serviced space ventilation noise, additional noise reduction treatment may be needed (see Section 4).

The fan generated sound power levels are first calculated as shown in Table 6.1 and Section 3.3. Then, for each serviced room considered (see Section 3.2), the calculation steps to obtain the fan noise and flow generated noise reaching the room are separately tabulated. Finally the calculations to obtain sound pressure levels within each room are tabulated. The details of these latter tabulated calculations are given in Sections 3.4, 3.5 and 3.6.

There are some steps detailed in the calculation procedure that obviously will not influence the sound pressure levels within the offices. For example, in the estimation of flow generated noise, the noise due to certain duct fittings does not influence the total airflow generated noise levels. After gaining some experience in using the Data Items on noise in air-conditioning systems, the user may be able to omit some of these calculations when considering low velocity systems.

3.2 Preliminary Assessment of the System

It is necessary to consider only the critical areas served by the ventilation system. In this case, the board room/conference room must be investigated because the required noise rating is the most severe in the office complex (see Tables 6.2, 6.3 and 6.4). Also the managing director's office should be considered because it is closer to the fan than the conference room and the flow control damper may generate significant levels of airflow generated noise (see Tables 6.5, 6.6 and 6.7). If the noise levels in the managing director's office and the conference room satisfy the required noise ratings, the noise levels within the other offices, that have higher permitted noise ratings, will be within the required limits.

A further point to consider in making the initial assessment of a system is the air flow rates to each room. Because the noise levels, due to the ventilation system, within a room are related to the air volume delivered to the room, consideration should be given to the sound pressure levels within a room if the air flow delivered to it, per unit volume of space, is significantly higher than that delivered to other rooms served by the duct branch.

3.3 Estimation of Fan Sound Power Levels

The fan sound power level in each octave band is estimated using the method given in Section 9.7 of ESDU 79037. However, if a specific fan has already been selected for the system it is recommended that the manufacturer's noise data are used.

The required volume flow through the fan, $q = 4.30 \text{ m}^3/\text{s}$.

The fan selected for this system is a centrifugal type with 32 forward-curved blades. It delivers the required volume flow when rotating at 680 rev/min and, at that operating condition, has a total pressure development, $\Delta p_{f,t} = 700 \text{ Pa}$.

The detailed calculation for the octave band centred at 250 Hz is given below, and values tabulated for each octave band in Table 6.1.

- 3.3.1** The base sound power level, L_w^* , is obtained from Figure 7 of ESDU 79037. For $q = 4.30 \text{ m}^3/\text{s}$ and $\Delta p_{f,t} = 700 \text{ Pa}$, $L_w^* = 49 \text{ dB}$.
- 3.3.2** The specific fan sound power level, K_w , is obtained from Table 9.2 of ESDU 79037 for the appropriate fan type. For the 250 Hz band for a centrifugal fan with forward-curved blades, $K_w = 36 \text{ dB}$.
- 3.3.3** From Figure 6 of ESDU 79037, for a fan with 32 blades rotating at 680 rev/min, the blade passing frequency, f_B , falls in the octave band centred at 500 Hz. From Table 9.2 of the same Item, the blade frequency increment is 2 dB.

3.3.4 The total in-duct sound power level, L_w , in each octave band is given by

$$L_w = L_w^* + K_w + \text{blade frequency increment.}$$

For the 250 Hz octave band,

$$L_w = 49 + 36 + 0 = 85 \text{ dB.}$$

3.4 Propagation of Fan Noise Through the System

The calculations for the propagation of fan noise through the system are tabulated for each of the eight octave bands of interest in Tables 6.2 and 6.5 for the board room/conference room and the managing director's office branches, respectively. Details of these tabulated calculations, using for illustrative purposes (where appropriate) data for the single octave band centred at 250 Hz, follow.

3.4.1 For the 8 m length of unlined duct, between the fan and branch A, of cross section $1000 \times 500 \text{ mm}^2$,

$$\frac{P}{A} = \frac{2(1.0 + 0.50)}{1.0 \times 0.50} = 6.0 \text{ m}^{-1}.$$

From Table 10.1 of ESDU 81043, the sound attenuation in the duct in the 250 Hz band is $8.0 \times 0.3 = 2 \text{ dB}$.

3.4.2 The attenuation* of sound power at duct branch A, in the duct considered, is given by Equation (3.5) of ESDU 81043.

$$\begin{aligned} \Delta L_w &= 10 \log_{10} \left\{ \frac{\text{sum of cross-sectional areas of ducts after branch}}{\text{cross-sectional area of duct}} \right\} \\ &= 10 \log_{10} \left\{ \frac{(0.65 \times 0.50) + (0.65 \times 0.50)}{(0.65 \times 0.50)} \right\} = 3 \text{ dB.} \end{aligned}$$

3.4.3 For the unlined duct, between A and B, of cross section $650 \times 500 \text{ mm}^2$,

$$\frac{P}{A} = \frac{2(0.65 + 0.50)}{0.65 \times 0.50} = 7.1 \text{ m}^{-1}.$$

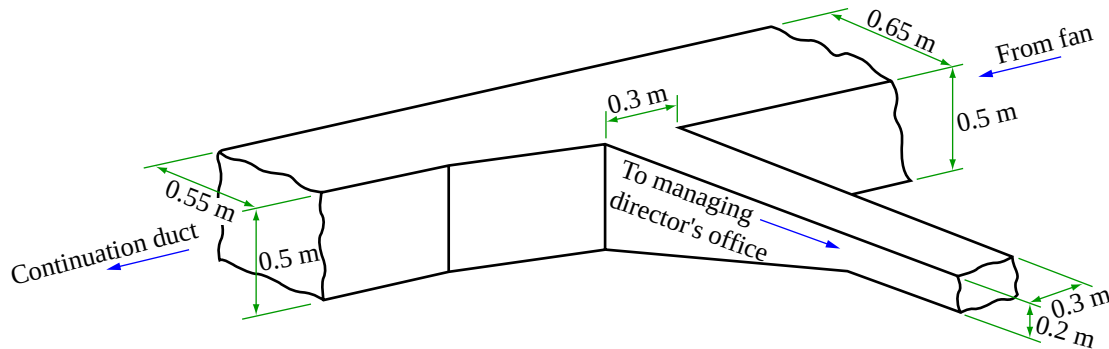
From Table 10.1 of ESDU 81043, the sound attenuation in a 4 m length of duct in the 250 Hz band is $4 \times 0.3 = 1 \text{ dB}$.

3.4.4 The attenuation of sound through the bend with turning vanes at B is given in Table 10.3 of ESDU 81043. For a duct dimension in the plane of the bend, d_c , of 0.65 m, the attenuation in the octave band centred at 250 Hz is 3 dB.

3.4.5 The attenuation of sound in a 1.5 m length of the duct considered in Section 3.4.3 is obtained from Table 10.1 of ESDU 81043. In the octave band centred at 250 Hz the attenuation is negligible.

* For a T-section branch, such as that at location A, the attenuation of sound given by Equation (3.5) of ESDU 81043 is conservative since it does not account for the sound energy that is reflected back towards the fan.

- 3.4.6** The sound attenuation at branch C may be calculated using Equation (3.5) of ESDU 81043. However, because of the difficulty in assessing the effective duct cross-sectional areas at the transition section, an alternative approach is used here based on volume flow rates. The details of the ductwork at branch C are shown in Sketch 3.1.



Sketch 3.1 Duct branch C with transition section in branch to managing director's office (see Section 4.2 on duct design)

As the volume flow rates are not given on Figure 1, they may be estimated using the relationship,

$$\text{volume flow rate} = \text{flow velocity} \times \text{duct cross-sectional area}.$$

$$\text{Volume flow before branch C} = 6.62 \times 0.65 \times 0.50$$

$$= 2.152 \text{ m}^3/\text{s}.$$

Volume flow towards the managing director's office

$$= 4.78 \times 0.30 \times 0.20$$

$$= 0.287 \text{ m}^3/\text{s}.$$

Therefore, volume flow in the main duct after branch C

$$= 2.152 - 0.287$$

$$= 1.865 \text{ m}^3/\text{s}.$$

The octave band attenuation, ΔL_w , is given by,

$$\Delta L_w = 10 \log_{10} \left\{ \frac{\text{total volume flow rate before the branch}}{\text{volume flow rate in branch duct considered}} \right\}. \quad (3.1)$$

Sound attenuation, at branch C, in the branch leading to the managing director's office

$$= 10 \log_{10} \left\{ \frac{2.152}{0.287} \right\} = 9 \text{ dB}.$$

Sound attenuation, at branch C, in the continuation branch

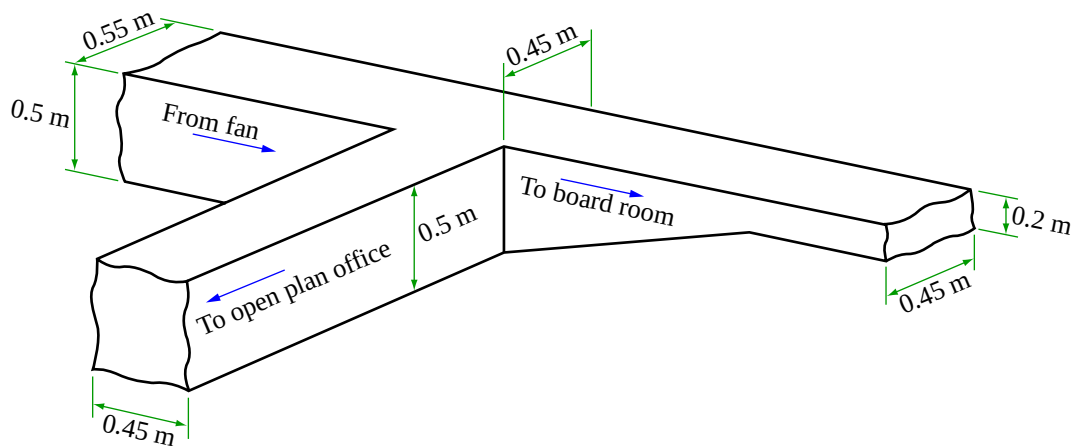
$$= 10 \log_{10} \left\{ \frac{2.152}{1.865} \right\} = 1 \text{ dB.}$$

3.4.7 For the unlined duct, between C and G, of cross section $550 \times 500 \text{ mm}^2$,

$$\frac{P}{A} = \frac{2(0.55 + 0.50)}{0.55 \times 0.50} = 7.6 \text{ m}^{-1}.$$

From Table 10.1 of ESDU 81043, the sound attenuation in a 6.3 m length of duct in the 250 Hz band is $6.3 \times 0.3 = 2 \text{ dB}$.

3.4.8 The sound attenuation at branch G is calculated in a similar manner to that used for branch C in Section 3.4.6. The details of the ductwork at branch G are shown in Sketch 3.2.



Sketch 3.2 Duct branch G with transition section in branch to board/conference room (see Section 4.2 on duct design)

Volume flow before branch G

$$\begin{aligned} &= 6.44 \times 0.55 \times 0.50 \\ &= 1.771 \text{ m}^3/\text{s.} \end{aligned}$$

Volume flow towards open plan office

$$\begin{aligned} &= 5.54 \times 0.50 \times 0.45 \\ &= 1.247 \text{ m}^3/\text{s.} \end{aligned}$$

Therefore, volume flow towards conference room

$$\begin{aligned} &= 1.771 - 1.247 \\ &= 0.524 \text{ m}^3/\text{s.} \end{aligned}$$

Using Equation (3.1), the sound attenuation in the conference room branch is given by

$$\Delta L_w = 10 \log_{10} \left\{ \frac{1.771}{0.524} \right\} = 5 \text{ dB.}$$

3.4.9 For the unlined duct, between G and H, of cross section $450 \times 200 \text{ mm}^2$,

$$\frac{P}{A} = \frac{2(0.45 + 0.20)}{0.45 \times 0.20} = 14.4 \text{ m}^{-1}.$$

From Table 10.1 of ESDU 81043, the sound attenuation in a 2.1 m length of duct in the 250 Hz band is $2.1 \times 0.3 = 1 \text{ dB}$.

3.4.10 The attenuation of sound through the bend at H is given in Table 10.3 of ESDU 81043. For a duct dimension in the plane of the bend, d_c , of 0.45 m, interpolation gives the attenuation in the octave band centred at 250 Hz as 4 dB.

3.4.11 The duct between H and I is of the same dimensions as the duct considered in Section 3.4.9. The attenuation of sound in 2.9 m length is obtained from Table 10.1 of ESDU 81043. In the octave band centred at 250 Hz, the attenuation is $2.9 \times 0.3 = 1 \text{ dB}$.

3.4.12 The attenuation of sound at branch I is given by Equation (3.5) of ESDU 81043,

$$\Delta L_w = 10 \log_{10} \left\{ \frac{(0.25 \times 0.20) + (0.25 \times 0.20)}{(0.25 \times 0.20)} \right\} = 3 \text{ dB.}$$

Note: The duct length between branches I and J is too short to provide significant sound attenuation. The two outlets fed by branch J will be the noisiest outlets into the board room due to the shorter duct lengths to those outlets.

3.4.13 The attenuation of sound at branch J is given by Equation (3.5) of ESDU 81043,

$$\Delta L_w = 10 \log_{10} \left\{ \frac{(0.20 \times 0.20) + (0.20 \times 0.20)}{(0.20 \times 0.20)} \right\} = 3 \text{ dB.}$$

3.4.14 For the unlined duct, of cross section $200 \times 200 \text{ mm}^2$, between branch J and the exit grille,

$$\frac{P}{A} = \frac{2(0.20 + 0.20)}{0.20 \times 0.20} = 20 \text{ m}^{-1}.$$

From Table 10.1 of ESDU 81043, the sound attenuation in a 1.1 m length of the duct is obtained. In the octave band centred at 250 Hz, the attenuation is negligible.

3.4.15 The outlet area at each outlet into the board room is 0.040 m^2 . The sound attenuation at the duct outlet is obtained from Figure 3 of ESDU 81043. In the octave band centred at 250 Hz, the outlet attenuation is 5 dB.

- 3.4.16** The sound attenuation at branch D is estimated using Equation (3.5) of ESDU 81043. For the branch leading to the managing director's office,

$$\Delta L_w = 10 \log_{10} \left\{ \frac{(0.20 \times 0.20) + (0.20 \times 0.20)}{(0.20 \times 0.20)} \right\} = 3 \text{ dB.}$$

- 3.4.17** For the unlined duct, of cross section $200 \times 200 \text{ mm}^2$, between locations D and E,

$$\frac{P}{A} = \frac{2(0.20 + 0.20)}{0.20 \times 0.20} = 20 \text{ m}^{-1}.$$

From Table 10.1 of ESDU 81043, the sound attenuation in the 2.8 m length of duct between locations D and E in the 250 Hz band is $2.8 \times 0.3 = 1 \text{ dB}$.

- 3.4.18** The sound attenuation at bend E is given in Table 10.3 of ESDU 81043. For this bend, $d_c = 0.20 \text{ m}$ and the attenuation is found by interpolation between the tabulated values for each octave frequency band. For the octave band centred at 250 Hz, the attenuation is negligible.
- 3.4.19** The sound attenuation in the 1.2 m length of unlined duct between E and F is found as in Section 3.4.17. For the octave band centred at 250 Hz, this attenuation is negligible.
- 3.4.20** The attenuation of sound at duct branch F is given by Equation (3.5) of ESDU 81043,

$$\Delta L_w = 10 \log_{10} \left\{ \frac{(0.20 \times 0.15) + (0.20 \times 0.15)}{0.20 \times 0.15} \right\} = 3 \text{ dB.}$$

- 3.4.21** For the 1.2 m length of unlined duct between F and the grille serving the managing director's office,

$$\frac{P}{A} = \frac{2(0.20 + 0.15)}{0.20 \times 0.15} = 23.3 \text{ m}^{-1}.$$

The attenuation is obtained using Table 10.1 of ESDU 81043. For the octave band centred at 250 Hz, the attenuation is $1.2 \times 0.9 = 1 \text{ dB}$.

- 3.4.22** The outlet area at each outlet into the managing director's office is 0.030 m^2 . The attenuation at the duct outlet is obtained from Figure 3 of ESDU 81043. In the octave band centred at 250 Hz, the outlet attenuation is 6 dB.

3.5 Flow Generated Noise and its Propagation Through the System

The calculations of the flow generated noise and its propagation through the system are tabulated for each octave band in Tables 6.3 and 6.6 for the board room/conference room and the managing director's office, respectively. The details of these calculations are given in the following sub-sections and illustrated, where appropriate, by using data for the octave band centred at 250 Hz. The sound attenuations within the system, calculated for the fan noise in Tables 6.2 and 6.5, also apply to the flow generated noise. Flow generated sound power levels are estimated, in each octave band, using the relationship given in Equation (4.1) of ESDU 81043,

$$L_w = F + G + H.$$